Experimental Investigations on Cold Energy Storage Employing Heat Pipes for Data Center Energy Conservation

Masataka Mochizuki[1], Xiao Ping Wu[2], Koichi Mashiko[3], Yuji Saito[4], Randeep Singh[5]

[1,3,4,5]R&D Department, Thermal Technology Division, Fujikura Ltd, Tokyo, Japan;

[2]Fujikura America Inc., USA

stevewu@fujikura.com

Abstract- **A novel type of heat pipe application for cold energy storage has been proposed and discussed in this paper. The cold storage system is aiming at saving electricity for data center cooling. A typical wickless heat pipe thermosiphon (thermal-diode heat pipe) will be employed in this application. The thermosiphon cold energy storage systems can be designed into several types that are ice storage, cold water storage, and pre-cool heat exchanger. Those systems can be used for co-operating with conventional chiller system for cooling data centers. The heat load used for discussing in this paper is 8800 kW, which represents a large scale data center. The methodology addressed in this paper can be also converted into the middle and small sizes of the data centers. This type of storage system will help to downsize the chiller and decrease its running time that would be able to save significant electricity cost and decrease green house gas emissions from the electricity generation. The proposed systems can be easily connected to the existing conventional systems without major design changes. The analysis in this paper is using Air Freezing Index AFI >=400 °C-days/year for sizing the heat pipe modules. The locations where AFI has different value the storage size will be varied accordingly. The paper also addressed a result that an optimum size of cold energy storage system should be designed at a level to handle 60% of total yearly heat load of a data center.**

Keywords- Heat Pipe; Thermosiphon; Cold Energy Storage; Data Center; Power Consumption; Energy Saving

I. INTRODUCTION

Data center power consumption will be equivalent to 6.6 Mil. Houses power in 2011.The power consumption cost is one of the major operating costs in a data center. Since the power supplied to the data center is ultimately dissipated as heat, a significant fraction of total power consumption will be necessarily used for running cooling system. Kenneth (2006) investigated heat density trends in sever and storage products and charted projection from 2,000 W/m2 in the year 1992 increasing up to 60,000W/m2 in the 2010 (Kenneth, 2006).

The cooling power is estimated sharing 30% up to 50% of total power consumption in the data centers. Since the power is still increasing on yearly basis, the power consumption and cooling issues have already affected commercial market for server's applications. According to analysis, three out of ten organizations have been impacted by the power consumption to their plan to purchase servers. For example, a data center with 8800 kW heat load that could consume more than $4 million a year, which is just the electrical bill for the cooling. This would be a financial issue for a user who is planning to invest this kind of data center. Additionally, power consumption increase is directly relating to the greenhouse-gas emissions as analyzed by the scientists worldwide (Patel et al., 2002; Moore et al., 2004 and Schmidt et al., 2009).

In recent years, the global warming and abnormal weather like heat waves, coastal flooding, and glaciers melting have been reported and physically felt by humankind. To save our planet, consequently, investigations on energy conservation for major power consumers such as data centers are becoming an important issue.

In this paper, an innovative method, which is applying the thermosiphon (thermal-diode heat pipe) to capture cold energy from the ambient during cold weather and stores the cold energy in the ice or water phase, has been introduced. The storage system can be used for partially replacing conventional chiller system for the data center cooling to save significant electricity cost.

II. PRINCIPLE OF OPERATION

The proposed system utilizes a special type of heat pipe-wickless heat pipe or named thermosiphon as shown in Fig. 1.
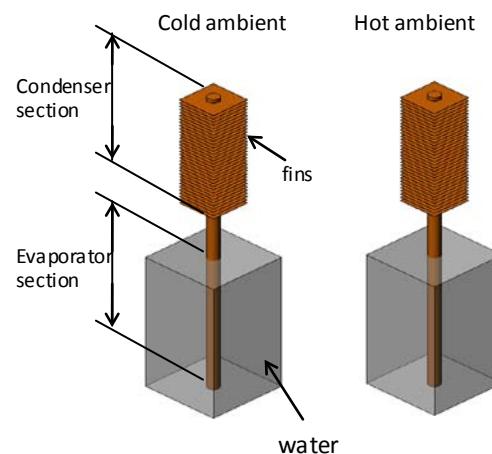

Fig. 1 Thermosiphon diode characteristics

A thermosiphon is made from a sealed metal pipe as its container. Unlike conventional heat pipes in which wick structure is built on the internal wall of the pipe, the thermosiphon internal wall has none wick to be built. As is well known, the function of the wick is for capillary of liquid from condenser section to evaporator section in the heat pipe and vaporizing the liquid when heat is applied at the heating area (Zhang et al., 2002 and Singh et al., 2009).

Without wick structure, the thermosiphon will not work when the heat is applied at the condenser section when this section is positioned on top of the thermosiphon, especially when the thermosiphon is vertically located. Thus,

alternatively, thermosiphon has another name,which is thermal-diode heat pipe.

A certain amount working fluid, which can vaporize and condense in a proper temperature range, will be charged into the pipe. All no-condensed gases inside the pipe will be fully evacuated. An evaporator section of the thermosiphon is locating at the bottom area of the pipe, which is dipped into the media (water) of the storage, and a condenser section with fins equipped is locating at the top area of the thermosiphon which will be exposed to the ambient air, as shown in Fig. 1.

When the media (water) temperature in the storage is higher than ambient temperature or the ambient temperature is lower than water temperature, for example, during a winter season in a cold territory, working fluid in the evaporator will be vaporized and the heat will be extracted from the high temperature media resulting in media temperature decreasing. The vapor will carry the heat up moving into the condenser section and transfer the heat into fins. The fins will dissipate the heat into the low temperature ambient air. Simultaneously, the vapor in the condenser section will be cooled down and condensed back to liquid phase then returning to the evaporator section by gravity. This thermal cycle will be continuously carried on by means of the evaporation-condensation process when the temperature of the storage media (water) is higher than the ambient temperature. However, when ambient temperature is equal to the storage media's temperature or higher, the heat transfer will be automatically stopped since thermal cycle has been broken due to none wick structure being built inside of the pipe. This is the unique characteristics of the thermosiphon (thermal diode heat pipe). With this one way heat transfer characteristics it can be ensured that the heat will not be transferred in opposite way from ambient to the media (water) during hot weather as shown in right hand side picture in the Fig. 1.

III. SYSTEM DESCRIPTION

Fig. 2 shows the schematic of the data center cooling system utilizing the proposed heat pipe based cold energy storage. The overall system consists of storage system 1, conventional chiller system 4 and data center server system 5.

In the storage system 1, the thermosiphons 2 are installed on top of the storage tank 3. Evaporator sections of the themosiphons will be dipped into the water media in the tank. The working relationship between storage system 1 and chiller system 4 is that storage system 1 will be simply connected with the chiller system 4 via pipes and three way valves. The three way valves can automatically control the water flow rate from storage system merging into the chiller system according to the water temperature variation in the storage tank. For example, during winter time, the water temperature is higher than ambient thus thermosiphons can extract cold energy from the ambient, the three way valve will allow more quantity of the water flow entering the chiller system resulting in that cooling load of chiller decreases, consequently, electricity consumption also decreases.

However, during warm season, when ambient temperature becomes higher than certain level of the water temperature, 25 °C for example, the thermosiphons will automatically stop working and the three way valves will be shut down. The storage system will be isolated from the chiller system and none water flow will enter into the chiller system from the

storage system. The heat load will be fully carried on by the chiller as a conventional system.

It is emphasized that the server system 5 and CPU temperature control will not be affected by the load changes year around from the cold energy storage system due to season variations. The cooling load running through that equipment will be the same as they are running in the conventional chiller systems. The only difference is that power consumption of chiller can be reduced since the storage system is involved.
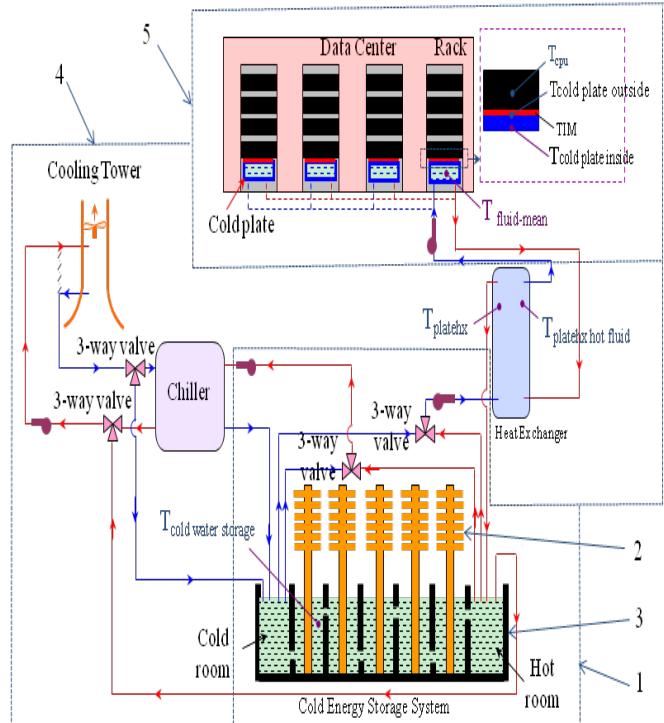

Fig.2 Cold storage system co-operating with chiller system: 1-cold storage system; 2- thermosiphon; 3- storage tank; 4 – chiller system; 5 – computer system

IV. THERMAL ANALYSIS

Fig. 3 shows geometry of the thermosiphon module to be discussed. The thermosiphon tube is made from stainless steel pipe. Fins are made from Aluminum. R134a is filled in the thermosiphon as a working fluid. The length of evaporator section is 3 m and condenser is 2 m respectively.
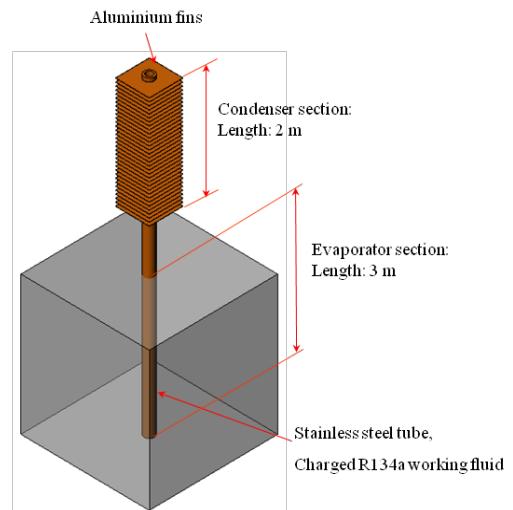

Fig. 3 Geometry of thermosiphon module

For the design of the thermosiphon cold energy storage system, firstly, water calculation temperature in the storage tank is one of the most essential parameters to be determined. This water temperature should be sufficiently lower than CPU surface temperature being able to cover the temperature rising range that caused by thermal resistances from heat transfer equipment as heat exchanger (estimated as 15 °C), contact thermal resistance as heat transfer through layer(s) of thermal interface material (TIM) (estimated as 15 °C) and heat gained from piping surface during transportation of the water (estimated as 5°C) as well as other unpredicted cold energy loss (estimated as 5°C). Thus, total temperature rise would be 40°C. Consider the nominally permissible temperature for the CPU surface T_{cpu} is usually known as 65 °C. The maximum cold water temperature in the storage will be allowed as 65 – 40 = 25 °C. In another word, directly and fully using cold water from storage to cool the CPU, the maximum temperature of the water is supposed to be 25 °C. When storage water temperature is higher than 25 °C, the chiller will be fully taking part in the cooling operation and storage water system will be stopped.

Fig. 4 shows analysis of thermal resistance circuit for thermosiphon module. In the module, since water to thermosipon pipe outer wall is dominated by natural convection, the equation for calculating thermal resistance per unit length for this item is then listed as following:
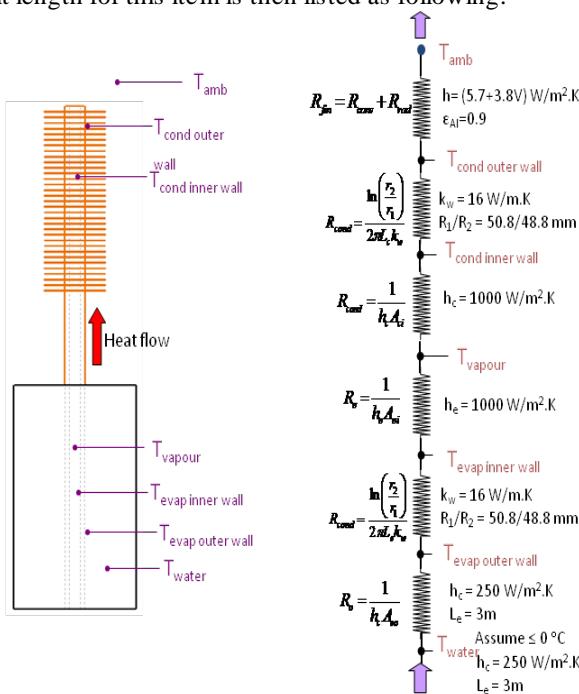

Fig. 4 Thermal resistance circuit from water through thermosiphon to ambient

$$R_o = \frac{1}{h_c A_{eo}} \quad (1)$$

The thermal resistance through pipe wall at evaporator section per unit length will be:

$$R_{cond} = \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi L_e k_w} \quad (2)$$

Then per unit length thermal resistance at evaporator section:

$$R_e = \frac{1}{h_e A_{ei}} \quad (3)$$

at condenser section internal wall surface:

$$R_{cod} = \frac{1}{h_c A_{ci}} \quad (4)$$

through condenser section wall:

$$R_{cond} = \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi L_c k_w} \quad (5)$$

and finally fins to the ambient:

$$R_{fin} = R_{conv} + R_{rad} \quad (6)$$

With the geometries shown in Fig. 3 and parameters shown in Fig. 4, the total thermal resistance through Equations (1) to (6) will be obtained as Rtl=0.02 °C/W for this module.

An experimental test module was built and tested at Aomoriwhere was locatingin a northern territory of Japan in 2008 and 2009. For investigating the cold energy capability that can be provided at a local area where the data center will be built, a parameter AFI is necessary to be introduced.AFI is defined as Air Freezing Index, which is given by the summation of the degree-days for freezing or thawing condition in the area interested.Equations (7) and (8) are the mathematical expression for AFI modeling:

$$\overline{T} = \frac{T_{min} + T_{max}}{2} \quad (7)$$

where T_{min} represents minimum daily air temperature and T_{max} represents maximum daily air temperature .

$$AFI = \sum_{i=1}^{N}\left(\overline{T_i}\right); \quad (8)$$
$$\overline{T_i} \leq 0$$

Table I shows some examples of calculation for AFI:

TABLE I EXAMPLE OF CALCULATION FOR AFI

Day	Max, °C	Min, °C	Average, °C	AFI, °C- Days
1	-1.70	-17.22	-9.46	-9.46
2	-12.80	-23.89	-18.34	-27.80
3	-12.20	-22.22	-17.21	-45.01
4	-9.44	-18.33	-13.89	-58.90
5	-1.10	-8.89	-4.99	-63.90
6	3.30	-1.10	1.10	-62.80
7	-1.10	-7.78	-4.44	-67.23

The result of final AFI will be adapted as an absolute value for record. For example, -67.23 °C-days in Table I will be recorded as 67.23 °C-days for convenience of applications. Table II gives AFI data for Poughkeepsie, New York, USA.

TABLE II AFI DATA FOR POUGHKEEPSIE,NEW YORK, USA

Description			Air Freezing Index Return Periods (ºF-Days) & Associated Probabilities (%)							
State and Station Name	Station Number	Mean Annual Temp. (º F)	1.1 Year (10%)	1.2 Year (20%)	2 Year (50%)	5 Year (80%)	10 Year (90%)	25 Year (96%)	50 Year (98%)	100 Year (99%)
POUGHKEEPSI E FAA AP	306820	49.1	265	355	554	769	885	1008	1088	1160
Description			Air Freezing Index Return Periods (ºC-Days) & Associated Probabilities (%)							
State and Station Name	Station Number	Mean Annual Temp. (º C)	1.1 Year (10%)	1.2 Year (20%)	2 Year (50%)	5 Year (80%)	10 Year (90%)	25 Year (96%)	50 Year (98%)	100 Year (99%)
POUGHKEEPSI E FAA AP	306820	9.5	147	197	308	427	492	560	604	644

* Upper table is sourced from National Climatic Data Center(NCDC) AFI value in lower table was converted from °F into °C by author. http://www.ncdc.noaa.gov/oa/ncdc.html

According to recommendation by NCDC, only the 100-year return period in this publication should be used, thus AFI

for Poughkeepsie is cited as 644°C-days. An experimental test rig was built in 2009 in Aomori in Japan and the tests had been carried out for two years during the winter time in 2009 and 2010.Fig. 5 shows the test resultsforJan 26 to March 26 in 2010.
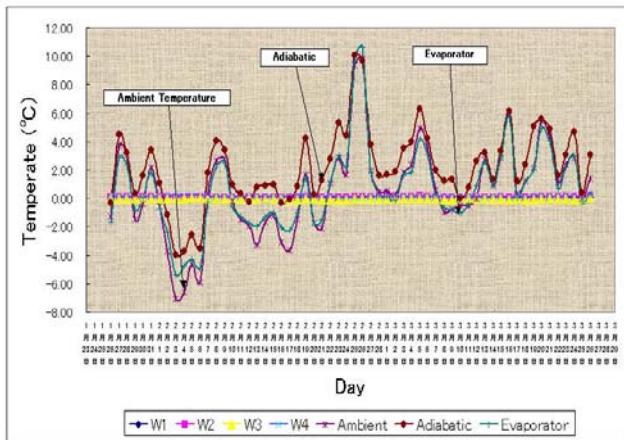

Fig. 5 Freezing and thaw data at Aomori Japan, 2010

In that period of time, the AFI was obtained as 56°C-days following the calculation method shown in Table Ⅰ.For a location where the air freezing index AFI is 644°C-days, the cold energy generated by this module can be estimated via calculation as shown in the following discussion. The average radius of ice column formed around the thermosiphon was 0.125m and height of the ice column was 2.6m in the test module while thermosiphon pipe outer radius is 0.025m. The water volume in the tank was about 10m^{3}which was measured before ice being formed. The average water temperature in the module tank was measured as 0.2 °C.

Fig. 6 and 7 show pictures of experimental rig and ice that was formed by the module tested. The total cold energy generated by one module will be analyzed as below:
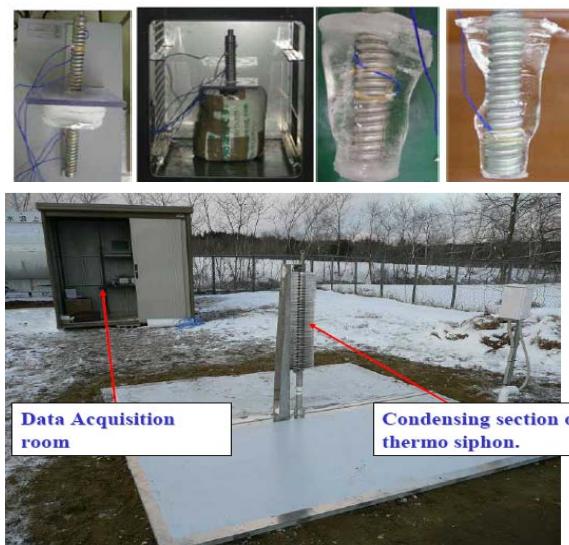
Fig. 6 Test rig of thermosiphon cold energy storage module
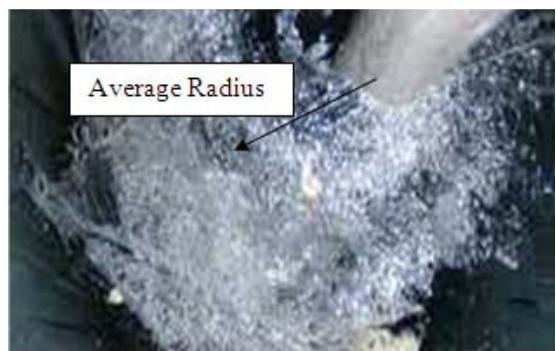
Fig. 7 Ice formed in the thermosiphon module

Total ice produced:

$$M_{ic} = 917kg/m^3 \text{ x } [\pi(0.125m)^2 x2.6m] – [\pi(0.0250m)^2 x2.6m]$$
$$= 113Kg.$$

Here 917kg/m3 is density of the ice.

The latent heat of the ice will be:

$$E_{lt} = 113kg \text{ x}334kJ/kg = 37.7MJ$$

Here 334kJ/kg is latent heat of the ice.

For a location with yearly freezing index 644°C-days,the module can produce latent cold energy per year as

$$37.7 \text{ MJ x } 644 °C\text{-days}/56°C\text{-days} = 434MJ.$$

As will be presented in Section Ⅴ, water will be heated up to 25 °C eventually, the sensible cold energy produced by the module per year will be

$$4200J/kg.°C \text{ x } 10m^3 \text{ x } 1000kg/m^3 \text{ x } (25 °C – 0.2 °C) = 1041 \text{ MJ}$$

Here 4200J/kg.°C is specific heat capacity of water and 1000kg/m^3 is density of water.

Thus total cold energy produced by the single module per year will be

$$434 \text{ MJ} + 1041 \text{ MJ} = 1475 \text{ MJ}.$$

For 8800KW heat load, assume 60% load (it will be further discussed in section 7.) or 5280KW (1665x10^5 MJ/year) will be carried out by the thermosiphon modules, the quantity of the modules needed will be 112880 units. For other percentage of heat load taking by thermosiphon the quantity of the modules will vary.Fig. 8 shows thermal net work for how cold water storage system is co-operating with chillersystem.
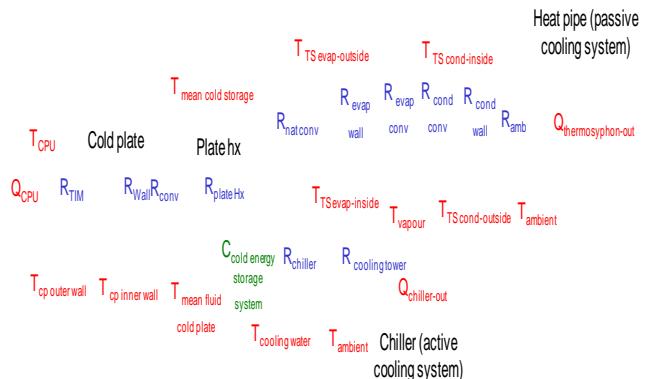

Fig. 8 Thermal net work of the entire cooling system: thermosiphon storage and chiller cooling system

From the net work, as we can see, heat generated from CPU will be dissipated to the ambient either through heat pipe

(thermosiphon) system or can also through chiller system. For a given heat load, if more load is carried on by the chiller system, there will be less load on thermosiphons.

V. SYSTEM DESIGN AND SIZING

The cold energy storage design and optimization size should be conducted on the basis of the local metrological conditions. Here Poughkeepsie, New York are taken for an example. The cold energy storage system is to be sized for a 8800 kW data center. The yearly temperature/wind data of the place, the number of thermosiphon modules and storage volume required for reducing chiller load by certain percentage will be discussed in this section. In the modeling, the payback time, which is used as the factor to optimize the cold energy storage size, will be also addressed.

Figs. 9 and 10 present the hourly ambient temperature data in °C and wind speed in m/s for Poughkeepsie in NY in the year 2008 respectively, which are used to determine the yearly heat transfer rate of the thermosiphon. The location has yearly average temperature of 10 °C and yearly average wind speed of 1.68 m/s as well as the freezing index of approximately 644 °C-days throughout the year.
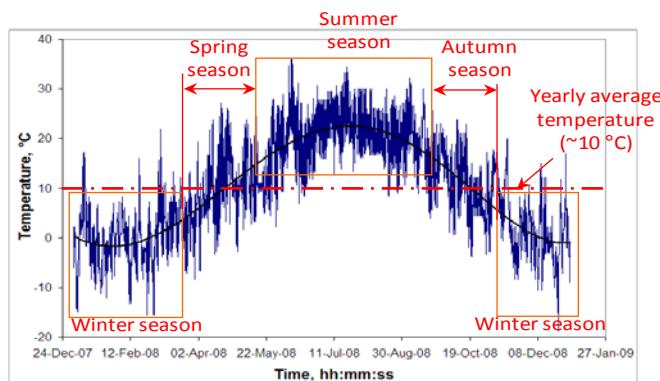

Fig. 9 Hourly temperature variation, Poughkeepsie, NY, 2008
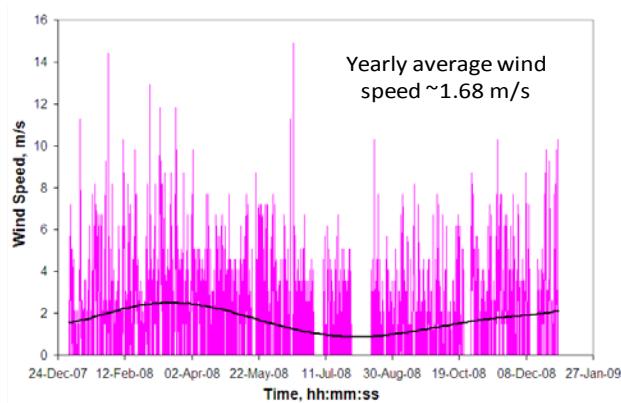
Fig. 10 Hourly wind velocity variation, Poughkeepsie, NY, 2008

Generally speaking, the temperature of the water in the storage will be changing seasonally as shown in Fig. 11. From January to March and October to December, the water temperature is relatively low. 100% cold storage water can be considered being used for cooling CPU. In this period, the CPU temperature would be varied within an acceptable range from 55°C to 65 °C as shown in Fig. 11. The chiller can also be run in this period to help to control the CPU temperature at stable level, but this will not be a point for discussion here in this paper.
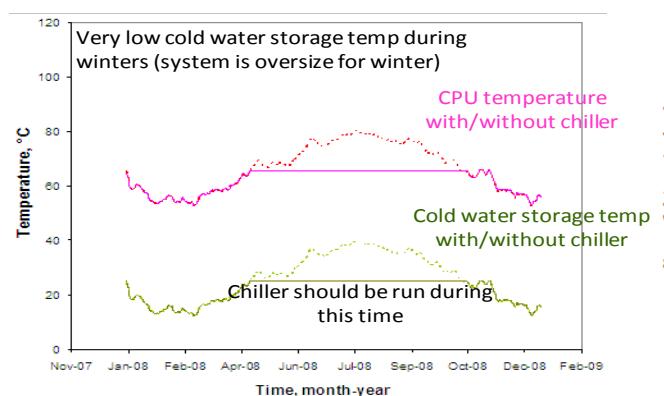

Fig. 11 Yearly variation in the CPU and cold water storage temperatures with and without chiller

As soon as the storage water temperature becomes higher in other warm seasons, the chiller will run and take more and more cooling load from the storage system till the thermosiphon fully stop. This would be a gradual process.

In Fig. 11, the lower curve represents storage water temperature variation. The solid line sections of the curve corresponding to the time from January to early April, early October to end of December show water temperature varies from 15°C to 25 °C. In this period, storage water could handle 100% cooling load. The size of the storage tank and number of the thermosiphon modules should be selected to meet this requirement of thermosiphon storage system design. The dot line in the same lower curve, which represents storage water, becomes higher than 25 °C in the warm remaining seasons. In this period, the thermosiphons will stop working and the chiller should run.

In Fig. 11 the upper curve represents temperature variation in CPU versus the lower cure (storage). It also shows chiller should run during the warm seasons. However, running about six months with the cold storage system will save significant power. Fig. 12 shows the estimated year round load share ratios in percentage between storage system and chiller system.
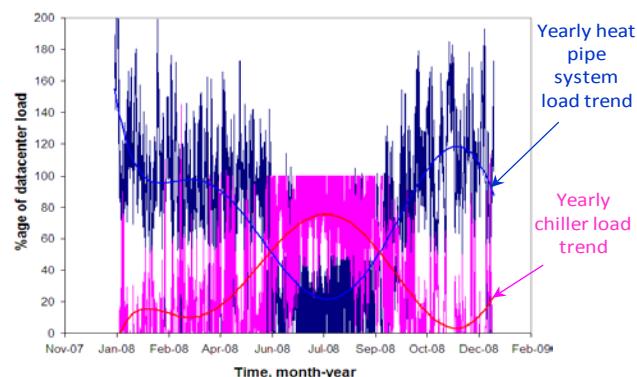

Fig. 12 Percentage of load handled by cold water storage and chiller throughout the year

The size of the thermosiphon is optimized on the basis of the payback time, which is defined as the ratio of the total system cost to the annual savings. It is recognized employing a smaller size of a single module will result in larger quantity of the units to be required. In another words, the smaller size of thermosiphon is, the higher cost of unit price will be needed to handle the same load. It will come out that payback time for thermosiphon investment becomes longer. Based on

economical analysis, the 112,880 units as calculated in the following Section Ⅵ, is over sized in term of cost versus payback. The payback time will be up to five years in this size, which is not acceptable.

To reduce the payback time, the cooling capability of the single thermosiphon needs to be enhanced by means of increasing the heat transfer areas both on evaporator section and condenser section as well as fins. The computational analysis based on previous modeling and experimental data gives that length of the evaporator section will be increased to 4.5mm and condenser length with fins will be increased to 3mm as shown in Fig. 13.
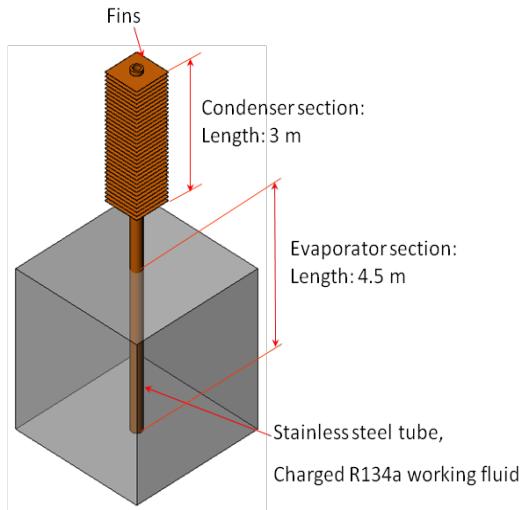
Fig.13 Enhanced thermosiphon module

Consequently, the 112,880 units will be reduced to 75,288 units. Following information will give the summary of related economical data. The unit cost of the thermosiphon will be controlled within $138 per unit. The storage volumetric space required is estimated as 15 m^3 per module, which will occupy 3.3 m^2 with 4.5 m depth. The chiller capital cost is constant for 8800 kW, which is counted as $500 per KW. In this analysis, the chiller is considered as being able to run at full load rate during peak summer time. And the electricity cost is estimated as $0.3 per KWHR.
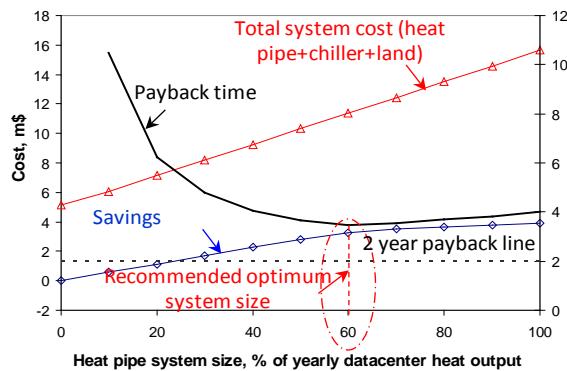
Fig. 14 Total storage system cost versus payback time

Fig. 14 plots the cost figures versus payback time for cold water storage system in which horizontal axle represents the percentage ratio for heat pipe (thermosiphon) storage system sharing the total cooling load. Left hand side vertical axle represents total cost, in million $, for the entire data center which includes storage system. Right hand side vertical axle represents payback time for the storage system. By the

analysis, 60% total cooling load being taken by cold storage is an optimum ratio. At this ratio, the electricity cost will be saved by approximate 3 million$/year. And the payback time is about 3.5 years (thermosiphon system cost versus savings). With the reduction of the power consumption about 10.4 kilotons of carbon dioxide will be reduced per year.

VI. OTHER TYPES OF HEAT PIPE BASED COOLING SYSTEM FOR DATA CENTERS

In this section, other types of heat pipe based cooling systems for power consumption saving would be briefly introduced.Fig. 15 shows a heat piped water pre-cooling heat exchanger co-operating with data center.
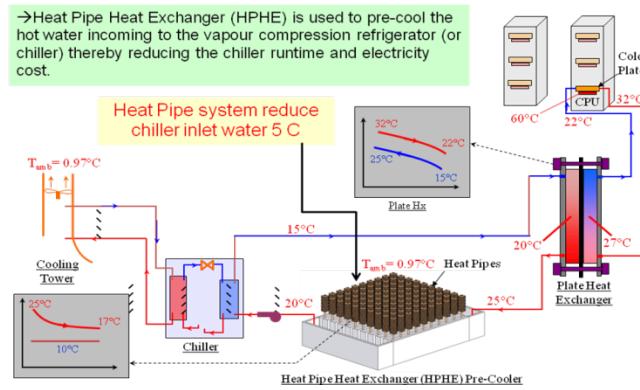
Fig. 15 Heat piped water pre-cooling heat exchanger

This system is employing a heat pipe array to act as a pre-cool heat exchanger to reduce the temperature of inlet water before entering the chiller. The cooling capability of the system is about ~5 °C down after the water going through the system. The energy saving can be estimated as 0.6 million$/year for 8800KW data centers.

Fig. 16 shows a heat piped ice storage system for data center power consumption saving system. This storage will be constructed underground with well insulated. It can be acting as a data center failure support system under an urgent situation when system power shut down. On the other hand,with enough amount of ice generated and stored, it could cool small scale data center year around without chiller. More discussions on heat pipe based pre-cool heat exchanger and ice storage systems will be further addressed in other papers.
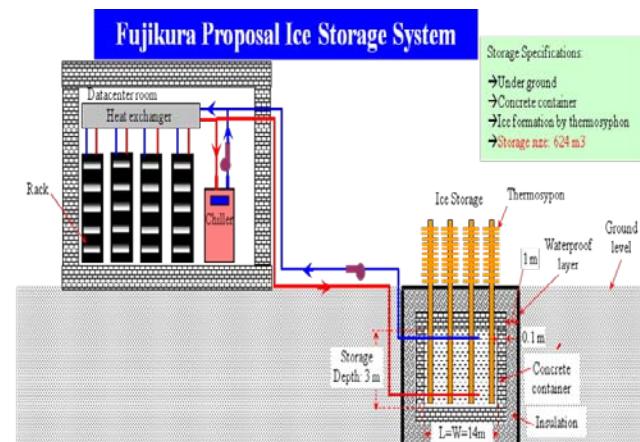
Fig. 16 Heat piped ice storage system

Another potential application of the heat pipe ice storage system is that,it could be a replacement of existing emergency power supply back-up system for data centers when power

supply fails for a short time, six hours for example. Fig. 17 shows a diagram for a type of existing data center emergency cooling system when power supply suddenly shuts down.
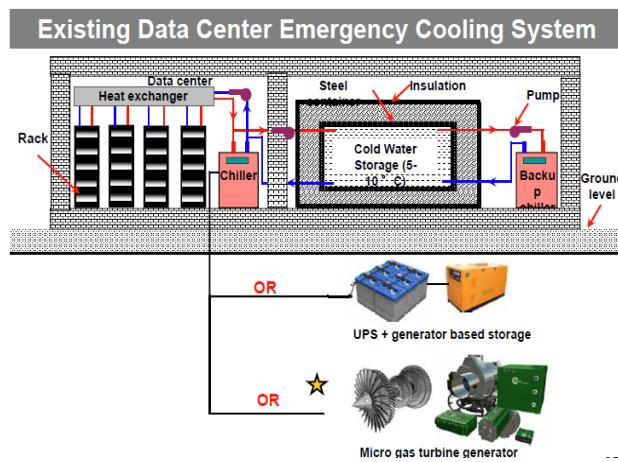

Fig. 17 Existing data center emergency cooling system

The existing systems employ UPS battery operated power generator or gas turbine operated generator to run cold water storage cooling system when emergency happens. The cold water storage consumes power during normal operation time even emergency dose not happening. However, heat pipe ice storage system(Fig. 16) will eliminate the traditional cold water storage system with zero power consumption during normal operations. When emergency happens, it only needs to pump the coolant through the ice storage system to get cooling energy to cool the data center.

VII.   CONCLUSION

In conclusion, the present paper has proposed a novel concept for thermal management of the data center using heat pipe technology. It can help to minimize the thermal load on the chiller units and thus save electricity as well as reducing green house gas emission. The system can be designed for servicing small and large size data centers. It also can be designed in various types with special functions for particular cooling applications.

However, it should be noticed that this type of cold storage is only recommended to build in the locations where cold seasons can provide sufficient freezing and thawing ambient temperatures. The air freezing index $A_I = 100$ °C - days is strongly recommended. Future investigations are still needed.

NOMENCLATURE

A – area, m2
E – energy, J
h – overall heat transfer coefficient, W/m2. K
K – thermal conductivity, W/m. K
L – length, m

r – radius, m
R – thermal resistance, °C/W
M – mass, kg
T – temperature, °C
$\overline{T}$ – mean daily temperature, °C
ε – emissivity

SUBSCRIPTS

amb – ambient
cpu – central processing unitor processor
cp – cold plate
c – convection, condenser
cod – condenser
conv – convection
cond – conduction
ci – condenser internal
ei – evaporator internal
eo – evaporator external
evap – evaporator
HX – heat exchanger
ic-ice
lt – latent heat
min – minimum
max – maximum
nat conv – natural convection
o – evaporator pipe outside
rad – radiation
tl – total
TS – thermosiphon

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